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Parameters of loading equilibration in a tube cold pilger mill drive

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Abstract On the basis of the variational approach and the Gauss-Seidel method there are proposed a technique and a mathematical model for determining the optimal parameters of dynamic load balancing systems on the crankshaft on the example of the cold rolling mill tube with reciprocating motion of an executive element in the form of a large mass working stand. The most compact scheme with the orthogonal motion of the executive element and the balancing load was chosen as the dynamic balance system. Variable parameters include dezaxial values, misalignment angle of cranks, weight of counterweight and balancing weight, lengths of connecting rods of executive and balancing mechanisms. For the existing series of sizes of cold rolling tube mill as the mass and speed of the rolling stand increase, the proportion of dynamic and technological components of the reduced load and respectively the kinematic scheme of the balancing mechanism changes. In this case, the structure of the loading and a set of variable parameters remain unchanged. Therefore, the proposed mathematical model of dynamic programming retains the universality of finding the minimum of maximum of the resulting load.

Cold pilger mills with rolls and with rollers are widely used for the production of precision cold-rolled tubes. The pilgering occurs in rolls, which have calibres, mounted in a massive working stand. At the same time the stand performs reciprocating motion. Along with a high strain of metal (up to 85%) by a full stroke the productivity of the process depends considerably on a stroke length, agility and feed motion of the workpiece over each cycle (a stand stroke). Mass of the stand and moving parts of her drive defines by rolling forces. Reciprocating motion of the working stand with pauses in end positions involves big accelerations and adequate to them resistance forces. Dynamic loads appeared in this is a speed restriction factor. For its dynamic equilibration there were engineered a number of designs [1]. However, the choice of the rational version for corresponding type of the cold pilger mill depends on main parameters, such as – a stand stroke length, moving parts masses, dimensional (linear and angular) characteristics of the main drive chains and balancers. The suggested study presents the general approach to the choice of rational parameters of the dynamic balancing system on the example of one of the widely spread kinematic scheme [2]. The total moment of the working stand main drive resistance (without balancing system) is characterized by a functional in which enters forces and moments of rolling, kinematic and mass characteristics of the drive system. Questions of experimental and computation definition of technological loads were given consideration in [3-6]. However, speed of cold pilger mills and, therefore, their productivity requires well-grounded choice of kinematic schemes and rational parameters of dynamic loading equilibration.

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The highest of numbers among presented loads has an unbalanced component coming from the periodic reciprocal motion of the stand. With a growth of speed the value gains defining value [7, 8]. Let's consider a perfectly rigid, anti-backlash system of the desaxial slider-crank mechanism.

Hereafter, let's consider possibilities of search for optimal parameters of the balancing mechanism of the cold pilger mill stand drive for shown below well-known version of such design (fig. 1).

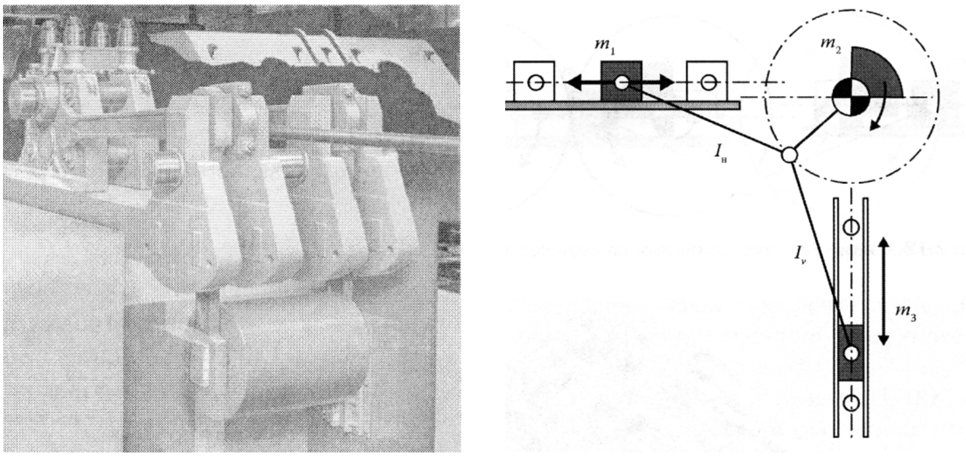

Figure 1. The CRP mill with the balancing mechanism (3D model).

The basis of the optimal parameters choice to minimize loading criteria is its functional and variation analysis of convenient functions those are brining minimum.

The functional of the system "drive-the working stand – balancing mechanism" loading presented graphically in normalized form in [1, 2].

Taking into account technological loading, which defines interaction between tools of the working stand and a rolled ingot, and also with a glance of moving masses and relation of master and slave links of the desaxial slider-crank mechanism, the expression for definition of torque modified to a crankshaft takes form of

$$T_M = F_G r K_D [\omega^2 r \frac{\cos\varphi + \lambda\cos 2\varphi + \zeta\sin\varphi}{g} + f +$$

$$+ \frac{n_Q F_{M\max}\sin^3\varphi}{F_G}](\sin\varphi + 0{,}5\lambda\sin 2\varphi - \zeta\cos\varphi) \quad (1)$$

Where r, ω – radius and angular velocity of crankshaft; φ – current angular coordinate of crankshaft; ζ, λ – desaxial relation and reverse relation of crankshaft length and crankshaft, respectfully; $n_Q F_{M\max}\sin^3\varphi$ – technological loading; m – modified mass of moving links, $m=F_G/g$; f, K_D – friction coefficient in kinematic pairs and dynamic factor, respectfully.

Therefore, the drive of the cold pilger mill stand (fig. 1) could be presented in a function of generalized coordinates depended in the end on the principal angular coordinate of crankshaft φ:

$$\upsilon = (\upsilon_1,\dots,\upsilon_n)^{\mathrm{T}} \quad (2)$$

Considering insignificant oscillations of angular velocity per the crankshaft turn $\omega=d\varphi/dt$ for the following analysis let's take her as constant value.

Let's divide the modified inertia moment to two components J_x and J_y – modified inertia moments of the stand and the balancing weight. With constant angular velocity of the crankshaft they give zero items. However their derivativesdJ_x and dJ_y don't equal to zero

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$$dJ_x=2\pi_{x\varphi}\pi_{x\varphi\varphi}m_x \tag{3}$$

$$dJ_y=2\pi_{y\varphi}\pi_{y\varphi\varphi}m_y \tag{4}$$

The static moment of balancing weights also don't equal zero

$$M_s=m_y\pi_{s\varphi} \tag{5}$$

$$M_k=m_{kr}\pi_{k\varphi} \tag{6}$$

The modified rolling moment and modified inertia moments of the stand and balancing weight presented as

$$M_o= M_{np}(x_{\kappa\pi})\pi_{x\varphi} \tag{7}$$

$$M_x=0.5dJ_x\omega^2 \tag{8}$$

$$My=0.5dJ_y\omega^2 \tag{9}$$

where

$$J_x=\pi^T{}_{x\varphi}\, m_x\, \pi_{x\varphi} \tag{10}$$

$$J_y=\pi^T{}_{y\varphi}\, m_y\, \pi_{y\varphi} \tag{11}$$

All listed components and the total moment $M=M_x+M_y+M_s+M_0+M_k$ shown graphically in fig. 3. In capacity of the functional let's take module of the total moment $|M|$.

The purpose of the optimization by Gauss-Seidel's method is a generalized choice of system parameters that delivers the minimum of maximized values of the total torque on the crankshaft a working cycle. A particular version of results of variation parameters influence for the crankshaft speed ω=12 s^{-1} shown in fig. 2. On Y-axis presented normalized values of torque those refer to loading in the unbalanced version of drive. On the X-axis there are normalized values of the parameter.

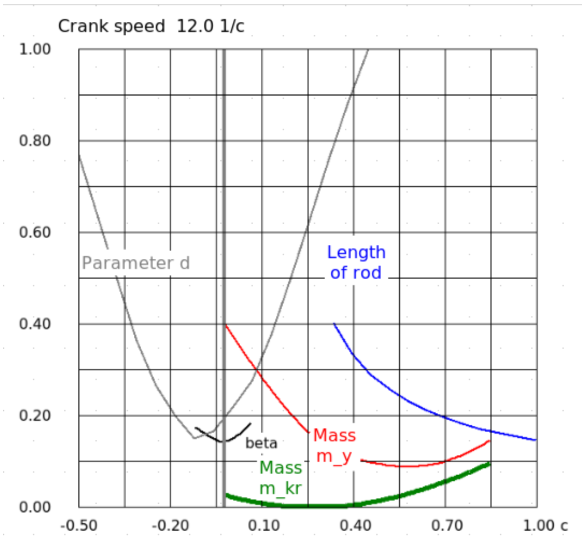
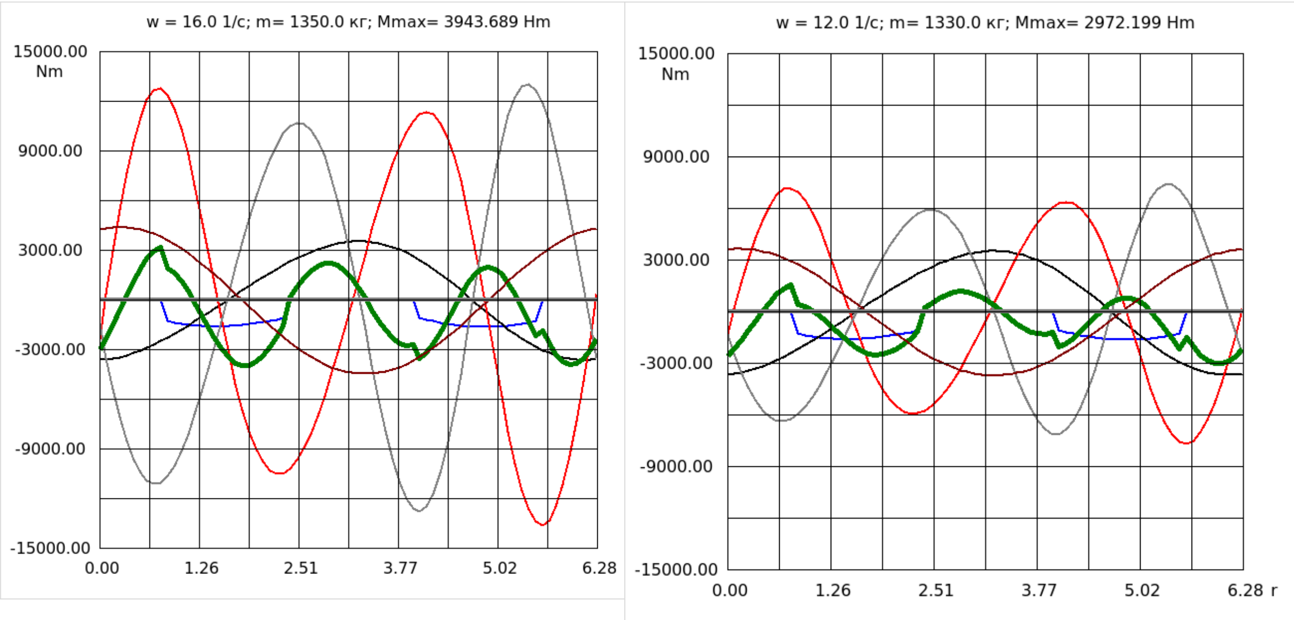


Figure 2. Optimal moments in relative units.

The search of minimax for different values of speed ω was carried out in the next sequence of parameters changing and their dependencies in the leverage system "desaxial –mismatching angle of the stand movement and the balancing weight crankshafts – the balancing weight mass – counterweight (weight on the crankshaft) mass”. For main types of the mill established that for middle and heavy types starting with a certain speed the value of balancing weight mass changes insignificantly. Mass of the weight perceptibly changes and reformatting of the functional defines by a configuration of the leverage system. Among parameters considered the most influential on resultant load is the balancing weight mass.

As the example, in fig. 3 versions of comparison between results of optimization of total moment on the crankshaft for the small cold pilger mill with the working stand mass 1350 kg and angular velocity of the crank ω=16 s^{-1} and ω=12 s^{-1}.



a b

Figure 3. Moments on the crankshaft (a)angular velocity of the crank ω=16 s^{-1}, (b) ω=12 s^{-1}: red - inertia moment of the stand, blue - inertia moment of the weight, brown - static moment of the weight, green - total moment.

During development of cold pilger mills with rolls and rollers driving mechanisms design with different loading balancing systems on the master shaft of the main additional restrictions are brought about by permissible loading on some parts and joints of the working line, including coupler, bearing, guides, etc. From this point of view comparable evaluation of possibility to increase of speed and performance of the mills from the choice of optimal balancing system parameters is of interest. For described particular version of the driving mechanism without balancing system safe loads in the main engine define recommended speed of crankshaft ω=12 s^{-1}. At the same time the crankshaft is undergoing of torque approximately 4.8 kNm. The choice of optimal parameters allows same loading with the increase of speed in 1.8 times.

Nowadays there are a number of different offers about loading equilibration in drives of the CRP from the main type row [1, 6]. The examined method of the choice of optimal versions of loading equilibration in known mechanisms has a general approach that could be apply in various kinematic schemes.

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Conclusions

A method of automated calculation of optimal parameters of loading equilibration for heavy-loaded drive mechanism of cold pilger mills with reciprocal motion of the working stand suggested. The method allows significantly increasing its speed and productivity on the basis of evaluation of influence on the equipment operability index.

The check of a dynamic mode showed insignificant difference from the examined case of static equilibration.

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